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AN INNOVATIVE METHOD FOR DETERMINING MAXIMUM POWER OBTAINABLE FROM A COMMON RAIL TURBOCHARGED DIESEL ENGINE STARTING FROM EXPERIMENTAL DATA

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ABSTRACT

This paper presents an experimental method for determining the maximum power obtainable from an aircraft diesel engine derived from the automotive field. Several tests were performed to estimate the real values of pressure and thermal loads on the piston and head. The simple and original method described in this paper made it possible to define two constants for the description of the engine-performance-envelope. Maximum allowable pressure on the piston and the head was evaluated with computer simulation. Two different indicator cycles of the same engine (in the automotive and aircraft version) were analyzed to extrapolate three constants that define the engine family:

- The contributing factor of compression (the exponent of the polytrophic curve n)
- The contributing factors of combustion (a constant and an exponent)

These contributing factors were examined and quantified separately.

The paper is composed of two parts: the first part highlights the ultimate loads on critical engine parts while the second details the method for the extrapolation of the ultimate power output value. Successive experimental tests performed on the engine confirmed the simple method described herein.

KEY WORDS

Diesel, pressure, power, engine.

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NOMENCLATURE

C	=	stroke;
D	=	bore;
k	=	constant;
n	=	polytrophic exponent;
n'	=	polytrophic exponent for the automotive version;
n_{comb}	=	number of burst per cycle;
P	=	power;
P_{max}	=	maximum obtainable power;
p	=	pressure;
p_1	=	manifold pressure;
p_2	=	pressure at the end of the compression;
p_{2max}	=	maximum pressure only due to the compression;
$p_{2'}$	=	absolute pressure relative to $-16.5°$ from the TDC (Top Dead Centre) for the automotive version;
p_{2max}'	=	maximum pressure only due to the compression for the automotive version;
q	=	fuel consumption x single burst;
$q_{Aeromin}$	=	fuel consumption per minute for the aeronautical engine version;
$q_{Automin}$	=	fuel consumption per minute for the automotive engine version;
q_{Aerosp}	=	specific fuel consumption for the aeronautical engine version;
q_{Autosp}	=	specific fuel consumption for the automotive engine version;
r	=	compression ratio;
r'	=	compression ratio for the automotive version;
V	=	displacement;
V_1	=	volume to be swept by the piston at the start of the compression
V_1'	=	V_1 calculated for the automotive version;
V_2	=	volume to be swept by the piston at the end of the compression;
V_2'	=	V_2 calculated for the automotive version;
$V_{unitary}$	=	unitary displacement;
V_{cc}	=	combustion chamber volume;
Δp	=	contribution of combustion to indicator cycle pressure for the aeronautical version;
$\Delta p'$	=	contribution of combustion to indicator cycle pressure for the automotive version;
Δp_{MP}	=	Δp referred to maximum obtainable power;
γ	=	numerical exponent used to calculate Δp;
ρ	=	inlet air density;
ω	=	crankshaft angular velocity;
ω'	=	reference crankshaft angular velocity for the automotive version.

INTRODUCTION

In the past there have been a number of attempts to produce diesel engines for aircraft; however, this technology has always had difficulty in handling the much higher compression ratios used in this type of design and suffered from poor power-to-weight ratios. Recent improvements in automotive diesel technology have lead to better power-weight ratios. In addition, state-of-the-art diesel engines enjoy high levels of fuel

efficiency, also in order to reduce emissions, [1-2], particularly when compared to gasoline engines and turboshafts.
Increased gasoline highly taxied fuel costs, compared to the JP4, JP8, and Jet A1 engines, have helped to revive interest in the diesel concept.
The power-to-weight ratio of diesels, specially when compared to turboprop engines, has, until now, prevented their use in large aircraft.
Diesel engines are more reliable and better suited to running for long periods of time at maximum power settings. Recent studies have been aimed at reaching the maximum obtainable power from a diesel engine.

DATA AND METHODS

To ascertain the maximum power obtainable from an aircraft diesel engine, it is necessary to estimate the real load on the piston and on the head. The engine's most stressed elements must sustain a real load made up of two different components: structural load due to pressure and thermal load.

The experimental method described in this paper is based on defining some constants that may be considered the same in similar engines. The first step is to evaluate the maximum acceptable pressure. To obtain pressure and temperature readings on the piston and on the head of the current engine, several tests were conducted on a two-litre diesel engine test bench in the engineering laboratory at Forlì II Engineering Faculty, Fig.1-2. The maximum allowable load on piston and head were obtained from FEA (Finite Elements Analysis) Fig. 3-4.

The FEA revealed that the piston ultimate load is 180 *bar* inside the combustion chamber and 160 *bar* on the piston top, Fig. 5. This is a significant result since it means it is possible to accept a higher level of pressure than what originally calculated by the piston manufacturer and consequently obtain more power. The FEA on the engine head resulted in structural limit at 260 *bar*, Fig.6. However, this analysis did not consider the increase in temperature due to the pressure increase; consequently this head structural limit may only be assessed as a theoretical value. Anyway it was possible to individuate the piston as the weak component of this engine.

Maximum allowable pressure is needed to determine maximum obtainable power. Pressure is assumed to the sum of two contributing processes: combustion and compression. These contributing values were analysed in view of the following hypotheses: compression can be considered as a polytrophic transformation [3] and the combustion follows an exponential low [4].

The basic assumption is that similar engines have in common the same transformations. Two engines are similar if they share the same combustion chamber and the same bore. Crankshaft speed should also be similar, it is not possible to use the same compression data for slow and high speed engines.

To study the compression process some characteristics of the engine have to be considered:

- ✓ manifold pressure
- ✓ inlet air density
- ✓ compression ratio for each engine

- ✓ unitary displacement
- ✓ combustion chamber volume
- ✓ the exponent of the polytrophic transformation (similar rotational speed)

As hypothesised, the compression phase follows a polytrophic law:

$$pV^{n} = \text{cost} \tag{1}$$

Our two-litre diesel engine data are:

- ✓ r = 15.5÷1
- ✓ V = 2000 *cm*^{3}
- ✓ C = 92.45 *mm*
- ✓ D = 83 *mm*
- ✓ ω = 3800 *rpm (aircraft version, 4250 automotive version)*

The polytrophic law can also be written as :

$$p_1(V_1)^n = p_2(V_2)^n \tag{2}$$

The unknown quantity is the exponent n. It is then possible to calculate n. Experimental data are available in the pressure-crank-angle diagram and power curve, Table 1, both for the aircraft and the automotive version of the engine. From the pressure-crank-angle diagram, the pressure value at each crank angle position is available. We assumed *n* as being constant during the whole transformation and we refer to two random pressure values during the compression phase. However it is better to chose two distant points of the indicator curve to improve precision. From the power curve data, the injection timing advance at 3800 *rpm* is -18° .

The other pressure value chosen is the pressure at the beginning of the compression on the power curve. Then we calculate the unitary displacement and the combustion chamber volume. Referring to V_1 as the combustion chamber volume plus unitary displacement and to V_2 as the combustion chamber volume plus the volume at α = -18° crank angle position, we can also calculate:

$$V_1 = V_{unitary} + V_{cc} = 53.2212 \times 10^{-5}\ m^3 \tag{3}$$

$$V_2 = V_{cc} + \frac{V_{unitary}}{C} \cdot \left(C - \frac{C}{2} - \frac{C}{2} \cdot \cos\left(\frac{\alpha}{180 \cdot \pi}\right)\right) = 4.44778 \times 10^{-5}\, m^3 \tag{4}$$

It is possible to calculate n:

$$n = 1.34 \tag{5}$$

Once this value is known, the pressure relative to combustion is obtained from the total pressure minus the maximum pressure due to compression. Maximum pressure alone due to compression can be evaluated by repeating the same process used to calculate n when p_1 and V_1 are the same as in the previous case and V_2 is the combustion chamber volume.

$$p_{2max} = 109.329 bar \tag{6}$$

The combustion phase follows the exponential law:

$$kq^{\gamma} = cost \quad (7)$$

The combustion process depends on the following factors:
- ✓ specific fuel consumption
- ✓ fuel consumption per cycle
- ✓ number of combustions per revolution
- ✓ crankshaft rotational speed
- ✓ the energy discharge process
- ✓ the cooling process

The contributing factor of combustion is calculated by subtracting the contributing factor of compression above calculated from the maximum pressure value, given by the pressure-crank-angle diagram.

Then we have to determine the k and γ values. We assume that these values are the same for similar engines, so we can refer to the automotive version data provided by the engine manufacturer. The manifold pressure, maximum pressure, and displacement value are the same as the engine test bench data. There are only two different values and they are:

- r´ = 18.5:1
- ω' = 4250 rpm

With a different compression ratio, the combustion chamber volume changes as well, and so V_1 , V_2, n and the final pressure alone due to the compression were recalculated, Fig.7.

We obtain the n' value in the same way as before but with the different pressure value relative to the different compression ratio.

$$n' = 1.27293 \quad (8)$$

Maximum pressure alone due to the compression is:

$$p_{2max}' = 110.891\ bar \quad (9)$$

where, as above, V_2 is to be considered V_{cc} relative to the new compression ratio.

The maximum pressure value is the same as in the previous case. From the fuel consumption for burst of both engines it is thus possible to determine k and γ.

In this example specific fuel consumptions are:

- q_{Aerosp} = 228.4 *g/kWh*;
- q_{Autosp} = 233.84 *g/kWh.*

It is now possible to estimate the fuel consumption for burst in both engines:

- n_{comb} = 2

The fuel consumption per burst is:

$$q_{Aeroburst} = \frac{q_{Aeromin}}{\omega \cdot n_{comb}} = \frac{0.418353}{3800 \cdot 2} = 0.00005504\,64\ \frac{kg}{burst} \tag{10}$$

$$q_{Autoburst} = \frac{q_{Automin}}{\omega \cdot n_{comb}} = \frac{0.328595}{4250 \cdot 2} = 0.00003865\,83\ \frac{kg}{burst} \tag{11}$$

It is possible to calculate k and γ:

$$\Delta p = k \cdot q_{Aeroburst}{}^{\gamma} \tag{12}$$

$$\Delta p' = k \cdot q_{Autoburst}{}^{\gamma} \tag{13}$$

$$k = 120.814\ \frac{bar}{kg/burst} \tag{14}$$

$$\gamma = 0.0885968 \tag{15}$$

DISCUSSION OF RESULTS

An example for the estimation of the maximum power obtainable in this two-litre piston engine is the main evaluation. By fixing the pressure value at the structural limit of the piston, which is 180 *bar*, it is possible to evaluate the maximum power. Writing q_{Aero} as a function of maximum power we can consider Δp as a function of maximum power. Calculating backwards it is possible to define the contributing factor alone due to the compression when maximum pressure and the Δp_{MP} are known.

The aeronautical version of the aftercooler can cool the air down to 338.15 K: from the assumption that under optimal conditions it can cool the air down to 323.15 K it is possible to calculate ρ at maximum power, manifold pressure at maximum power and Δp at maximum power, referred to the current power obtainable from the diesel engine test bench of 150 HP.

Maximum obtainable power = 185 HP

It is also possible to fix a required target maximum power and the maximum pressure in the combustion chamber in order to estimate the consequent compression ratio.
The final result is that at p_{max} = 180 *bar* and available power P = 200 *HP*:

r =14.5

Successive tests have demonstrated that the diesel test bench can sustain these power settings for long periods (hundred of hours). Engine performance at levels exceeding this power level have always led to rapid engine seizure and significant piston damage.

CONCLUSIONS

This experimental method has large fields of application; a spreadsheet is all that is required. The aim is to achieve optimum results in the simplest way possible. However, to obtain realistic data output, this method should only be applied using acceptable structural component limits. For industrial secret reasons, all numerical data does not refer to the original mentioned engine: the values the paper refers to have been adopted only for sample purpose.

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[4] Challen B., Baranescu R., Diesel engine reference book, Butterworth-Heinemann, (1999)

TABLES:

Table 1. Power curve

N TEST	*Rotational Speed*	*Load*	*Timing Advance*	*Specific Fuel Consumption*	*Manifold Pressure*	*Output Power*
	rpm	**%**	**deg**	**g/kWh**	**bar**	**kW**
1	4000	100	14.7	305.4	0.685	25.8
2	3900	100	17	229.3	1.43	79.2
3	3800	100	16.5	228.4	1.532	109.9
4	3500	100	14.7	222.8	1.528	107.4
5	3000	100	12.1	215	1.456	96.6
6	2500	100	11.5	213.8	1.201	76.7
7	2000	100	12	236.2	0.616	47.4
8	1750	100	10.1	231.3	0.26	30.6
9	1500	100	9.7	238.9	0.121	18.6

FIGURES

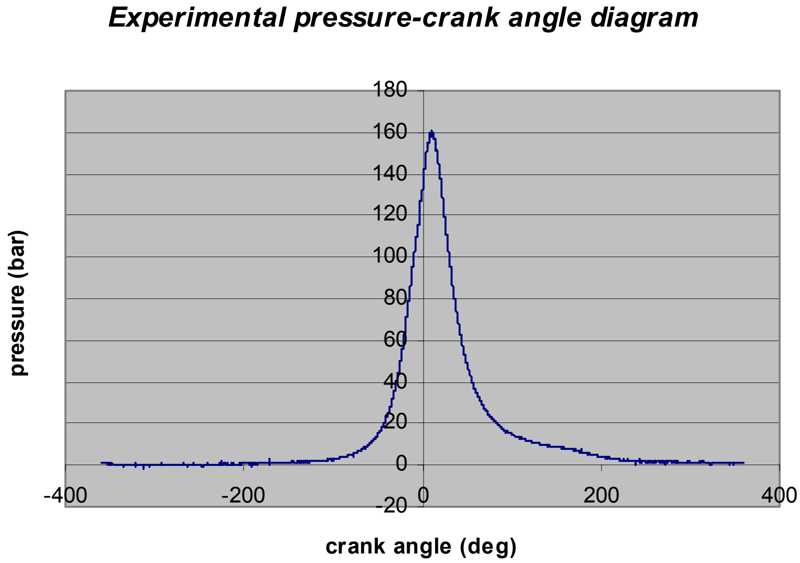

Fig.1. Pressure-crank-angle diagram at 3800 rpm (aircraft).

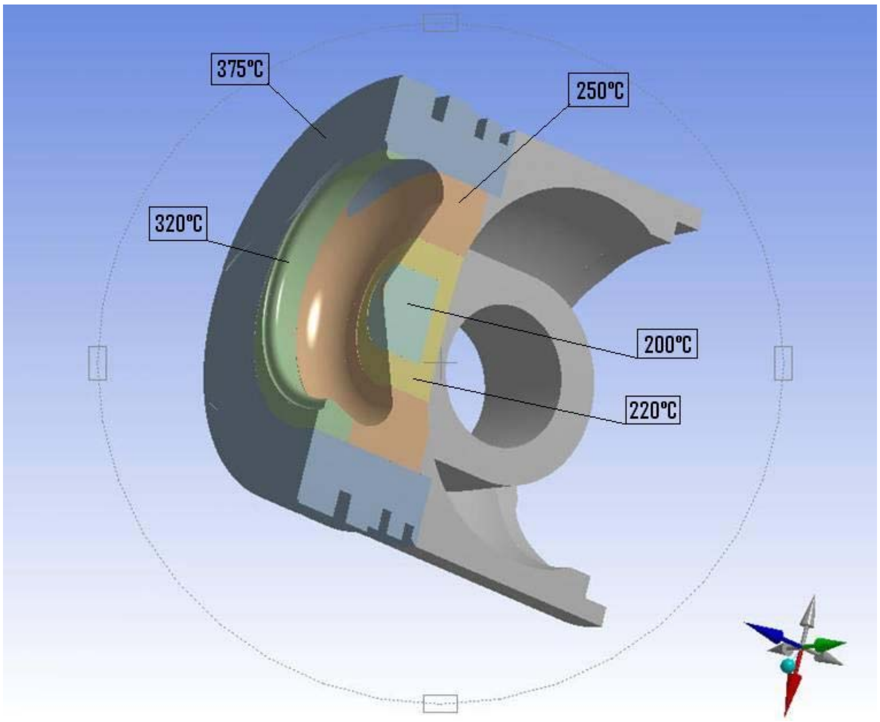

Fig.2. Piston temperature distribution map

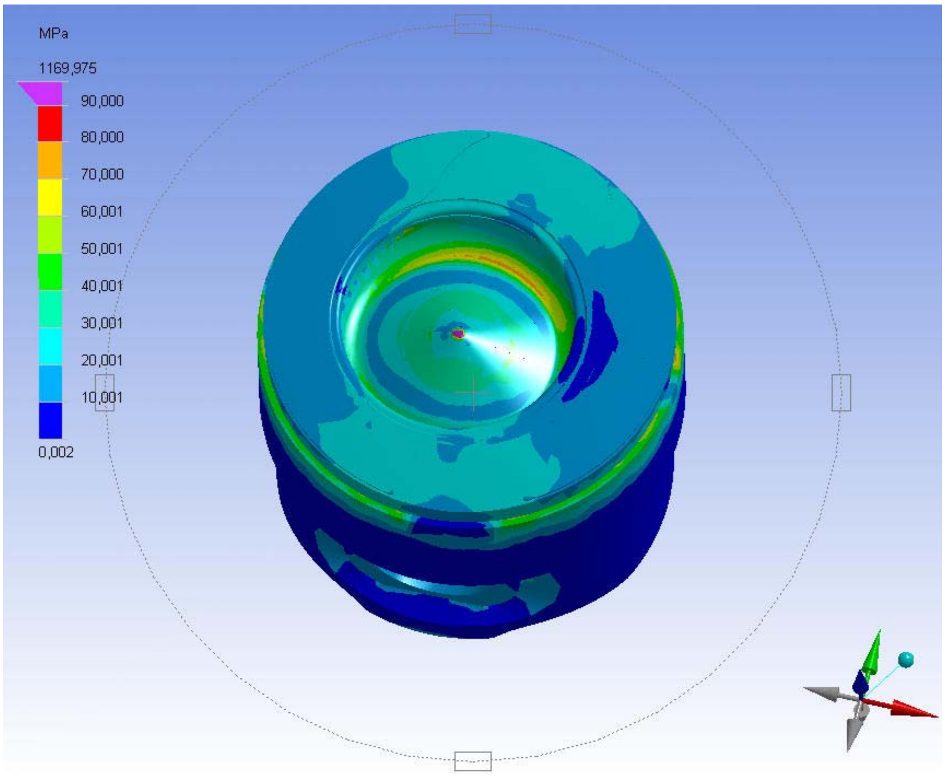

Fig.3. Piston stress under nominal conditions.

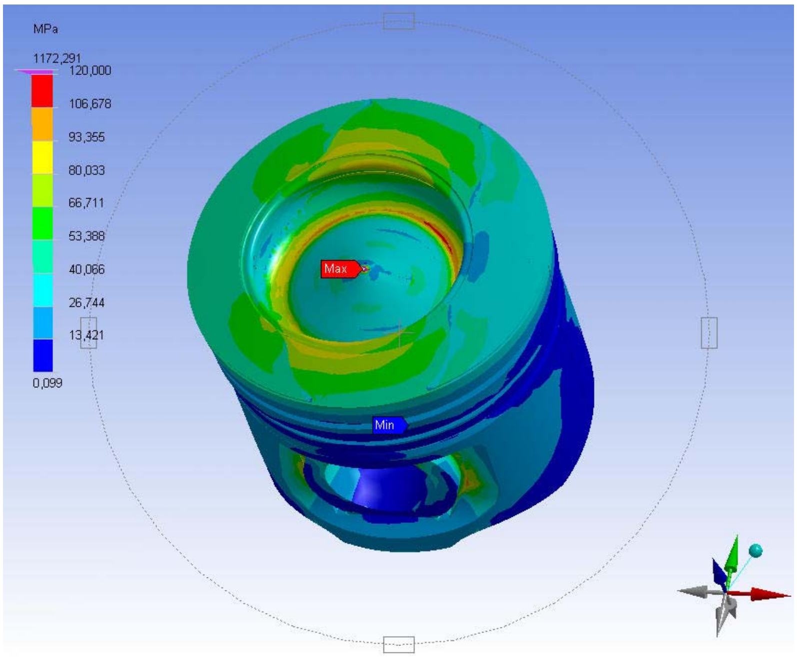

Fig. 4. The safety factor

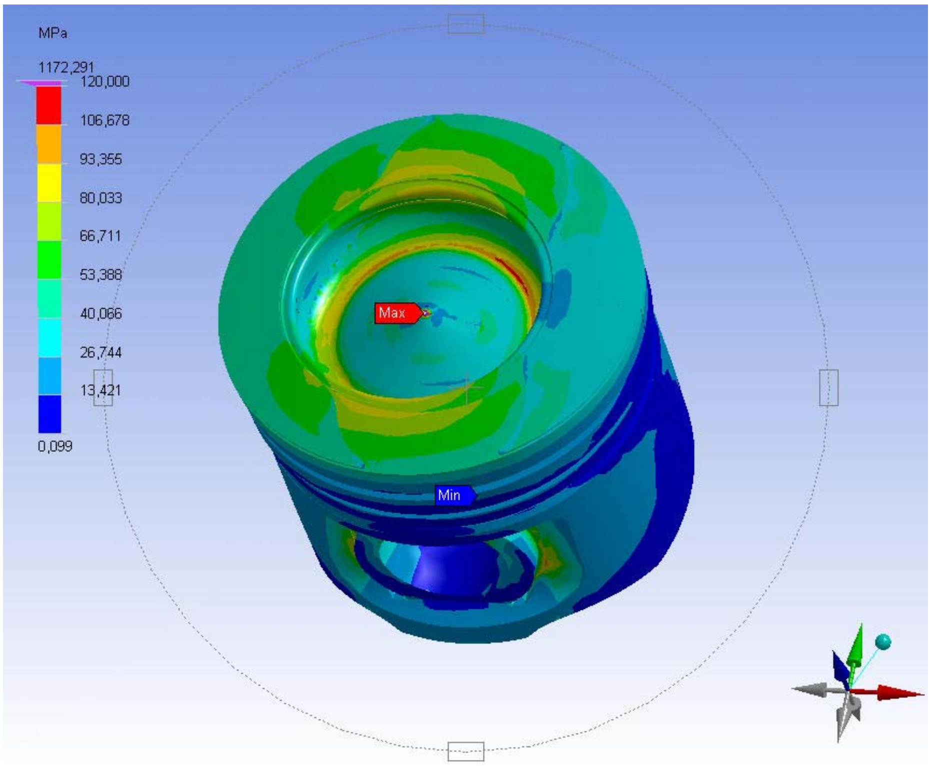

Fig.5. Piston structural limit.

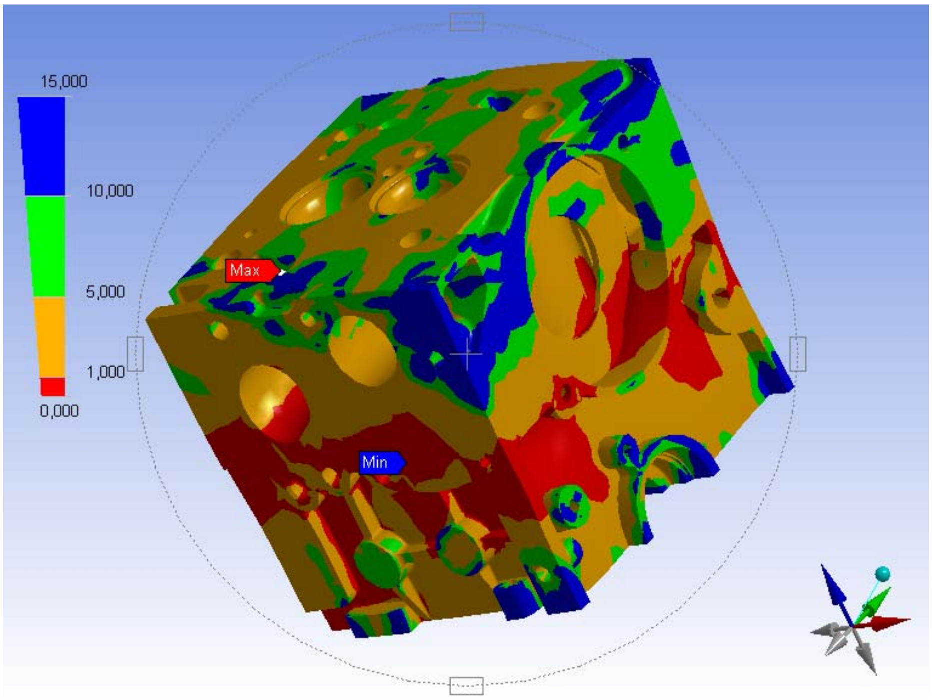

Fig.6. The head safety factor

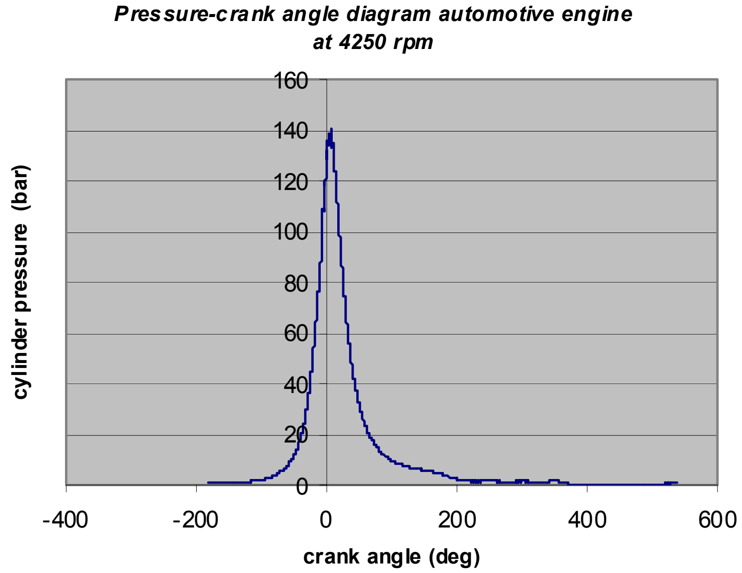

Fig.7. Pressure-crank angle diagram at 4250 rpm (automotive).